

# **Experimental investigation into the effect of magnetic fuel reforming on diesel combustion and emissions running on wheat germ and pine oil**

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## **ABSTRACT**

The present study aims to explore the effect of fuel ionisation on engine performance, emission and combustion characteristics of a twin cylinder compression ignition (CI) engine running on biofuel. Wheat germ oil (WGO) and pine oil (PO) have been identified as diesel fuel surrogates with high and low viscosities, respectively. High viscosity biofuels result in incomplete combustion due to poor atomisation and vaporisation which ultimately leads to insufficient fuel and air mixing to form a combustible mixture. Consequently, engines running on this type of fuel suffer from lower brake thermal efficiency (BTE) and higher soot emissions. In contrast, low viscosity biofuels exhibit superior combustion characteristics however they have a low cetane number which causes longer ignition delay and therefore higher NO emission. To overcome the limitations of both fuels, a fuel ionisation filter (FIF) with a permanent magnet is installed before the fuel pump which electrochemically ionises the fuel molecules and aids in quick dispersion of the ions. The engine used in this investigation is a twin cylinder tractor engine that runs at a constant speed of 1500 rpm. The engine was initially run on diesel to warm-up before switching to WGO and PO, this was mainly due to poor cold start performance characteristics of both fuels. At 100% load, brake thermal efficiency (BTE) for WGO is reduced by 4% compared to diesel and improved by 7% with FIF. In contrast, BTE for PO is 4% higher compared to diesel, however, FIF

has minimal effect on BTE when running on PO. Although, smoke, HC and CO emissions were higher for WGO compared to diesel, they were lower with FIF due to improved combustion. These emissions were consistently lower for PO due to superior combustion performance, mainly attributed to low viscosity of the fuel. However, NO emission for PO (1610 ppm) is higher compared to diesel (1580 ppm) at 100% load and reduced with FIF (1415 ppm). NO emission is reduced by approximately 12% for PO+FIF compared to PO. The results suggest that FIF has the potential to improve the combustion performance and reduce NO emission of high viscosity and low viscosity biofuels, respectively.

**Keywords:** Wheat germ oil; pine oil; fuel ionisation filter; biofuels; NO emission; electrochemical conversion

## Nomenclature

ATDC	After top dead centre	HC	Hydrocarbon
BP	Brake power	LVLC	Low viscosity and low cetane index
BSEC	Brake specific energy consumption	M	Methanol
BTDC	Before top dead centre	NO	Nitrous oxide
BTE	Brake thermal efficiency	PO	Pine oil
CAD	Crank angle degree	ppm	Parts per million
CI	Compression ignition	rpm	Revolutions per minute
CO	Carbon mono oxide	SCR	Selective catalytic reduction
CO <sub>2</sub>	Carbon dioxide	SOC	Start of combustion
DPF	Diesel particulate filter	TDC	Top dead center
EGT	Exhaust gas temperature	TFC	Total fuel consumption
FIF	Fuel ionisation filter	WGO	Wheat germ oil

## 1. Introduction

Global challenge facing automotive manufacturers is to develop an engine capable of meeting stringent emission standards without penalties in engine performance and driver experience. The latest aftertreatment technology to reduce NO and soot emissions are selective catalytic reduction

(SCR) and diesel particulate filter (DPF), respectively [1]. These aftertreatment devices significantly lower NO and soot emissions produced by compression ignition (CI) engines. However, both systems suffer from inherent limitations, SCR can only operate in a small temperature window and DPFs require regular regeneration which leads to penalties in fuel economy [2]. In addition, the capital, operational and maintenance costs are amongst the major drawbacks of both systems. The limitations in the available fossil fuel resources and the global concern over energy security has led to the development of low carbon and sustainable fuels as an alternative to diesel for CI engine applications. Vegetable oil is an attractive option due to lower emission levels and widespread availability. In particular, vegetable oil reduces the CO<sub>2</sub> emission, a significant contributor to global warming, since the plant feedstock absorbs CO<sub>2</sub> from the atmosphere during its growth [3].

Numerous vegetable oils including jatropha, karanja, neem and mahua have been previously investigated [4-7]. The experimental studies revealed that vegetable oils reduce HC, CO and CO<sub>2</sub> emission due to fuel-borne oxygen in their structure. However, the primary drawback of vegetable oil is its high viscosity and poor volatility which affects fuel atomisation, vaporisation and fuel-air mixing [8]. Therefore, use of vegetable oil in CI engines can lead to poor engine performance, piston ring sticking and injector clogging [9]. To overcome these issues, various strategies such as transesterification [10], fuel preheating [11] and use of hydrogen [12] and oxygenates [13] have been developed and tested by other researchers to improve the performance of vegetable oil in CI engines. Among these methods, preheating and transesterification alter the chemical/physical properties of the fuel prior to combustion whereas hydrogen and oxygenates are used as additives to improve the engine performance. Martin et al. [14] compared various methods to improve the performance of cottonseed oil in a single cylinder CI engine at full load condition. Additional methods such as transesterification, fuel preheating, DEE blending were studied with cottonseed oil as base fuel. They reported brake thermal efficiency improvement of 9% with ethyl ester as an oxygenate and 3% improvement with preheating at 110°C compared to base vegetable oil. Transesterification and fuel preheating led to increase in NO emission with reduced HC, CO and soot emissions compared to base vegetable oil. They concluded that among the tested methods, transesterification resulted in the maximum engine performance improvement.

Electrochemical conversion of fuel through magnetic ionisation has been tested in this study as a pre-combustion treatment system. Although limited studies have previously investigated the effect

of magnetic field on fuel atomisation, its effect on engine performance is yet to be fully explored. Fuel ionisation enhances the mixing of air and fuel molecules thus reducing fuel consumption and CO<sub>2</sub> emission [15-18]. Previous studies incorporated a permanent magnet in the fuel line/pump to reform the fuel prior to injection to improve the combustion. Patelet al. [15] and Okoronkwo et al. [16] studied the effect of the electromagnetic field on the ionisation and combustion of fuel in a single cylinder CI engine. The permanent magnets were placed in the fuel line with diesel as base fuel. They observed a significant reduction in harmful emissions such as HC, CO and soot. They also reported a decrease in fuel consumption due to improved mixing of ionised fuel molecules with air. Thiyagarajan et al. [18] conducted experiments with fuel magnetic ioniser in a single cylinder CI engine with ternary blends of karanja oil methyl ester, orange oil and methanol along with exhaust after-treatment system. They observed that ionisation of fuel blends aids in performance improvements resulting in higher brake thermal efficiency and reduced soot, HC and CO emissions. The results demonstrated that CO<sub>2</sub> emission could also be reduced with this technique.

In this study, wheat germ oil and pine oil with high and low viscosities, respectively, were utilised to assess the effect of magnetic ionisation on engine performance running on biofuel. Wheat germ oil is extracted from the germ of wheat kernels by cold pressing and supercritical CO<sub>2</sub> extraction [19]. Wheat germ is a by-product of wheat obtained from milling industries, which belongs to the Gramineae family [20]. Wheat germ oil is also used in cosmetics and for medicinal purposes. Wheat germ represents 2-3% of the entire wheat grain and it contains between 8% and 14% of oil [21]. Although the physiochemical properties of wheat germ oil suggest that it can serve as a diesel fuel surrogate, its potential as an alternative low carbon fuel in CI engines is yet to be explored.

Recently, researchers have changed focus to using fuels with lower viscosity and lower cetane index (LVLC) in CI engines due to improved fuel atomisation, evaporation and air/fuel mixing characteristics [22]. Various LVLC fuels such as eucalyptus oil, pine oil, camphor oil and orange oil have been tested [23]. Several researchers have investigated pine oil as a potential diesel surrogate fuel [24-26]. The calorific value of pine oil is comparable to diesel and hence making it suitable for CI engine operation without conversion to biodiesel. Vallinayagam et al. [24] examined the combustion and emission performance of a single cylinder, four stroke, direct injection diesel engine running on pine oil and its blends of 25%, 50% and 75% with diesel. The

experimental results indicated that pine oil has the potential to replace diesel in CI engines without any modification. The authors reported improved brake thermal efficiency and fuel consumption for pine oil compared to diesel at all load conditions. Brake thermal efficiency of pine oil was increased by 5% at full load condition. The maximum heat release rate of the engine at maximum power output was approximately 27% higher compared to diesel. Furthermore, CO, HC, and soot emissions were reduced with pine oil compared to diesel at higher load conditions. However, at full load, the results showed 25% increase in NO emission with pine oil compared to diesel.

The findings of the previous studies demonstrate that biofuel viscosity has an impact on engine performance and emission levels. High viscosity biofuels result in poor combustion due to inadequate fuel atomisation while low viscosity biofuels lead to increased NO emission and causes cold start issues due to low cetane index. Although the use of magnetic fuel ionisation as the pre-combustion system is a promising approach to tackle the above-mentioned shortcomings of high and low viscosity biofuels, limited research has been previously carried out by other researchers and it is yet to be fully explored. This study intends to ascertain the effect of magnetic fuel ionisation on engine performance and emission levels of a CI engine running on wheat germ oil and pine oil, high and low viscosity biofuels, respectively. The novelty of this study is the investigation of the effect of ion exchange in CI engines running on biofuel. Although pine oil has been previously explored as a potential diesel fuel surrogate, wheat germ oil is yet to be fully explored. This study aims to contribute to the knowledgebase by investigating the effect of magnetic ionisation on wheat germ oil in CI engines to overcome the fuel atomisation issues associated with this type of fuel, making it a viable low carbon alternative fuel for diesel-powered vehicles.

## **2. Methodology**

### *2.1. Test fuels*

In this study, wheat germ oil (WGO) and pine oil (PO) with high and low viscosities, respectively, were used as the test fuels while diesel was utilised as the base fuel. Table 1 shows the ASTM standards used to determine the fuel properties and Table 2 depicts the fuel properties of diesel, WGO and PO. It is evident that the viscosity of WGO is almost four times higher compared to diesel and the heating value is lower for WGO with similar cetane index compared to diesel. The

former causes poor fuel atomisation which leads to delayed start of combustion [27]. In contrast, PO has very low viscosity and similar heating value compared to diesel. However, cetane index of PO is markedly lower compared to the base fuel which results in longer ignition delay leading to higher NO emission and also causes cold start issues [25].

**Table 1**  
ASTM standards used for determination of fuel properties

Property	Test standards	Biodiesel standards		Method
		ASTM D6751 -02	EN 14214/14213	
Kinematic viscosity, cST @ 40°C	ASTM D445	1.9-6.0	3.5-5.0	Redwood Viscometer
Density @ 15°C, g/cm <sup>3</sup>	ASTM D1298	-	0.860-0.900	Pycnometer
Lower Heating value kJ/kg	ASTM D240	-	Min 35000	Bomb calorimeter
Cetane index	ASTM D976	Min 47	Min 51	Calculated based on API gravity & mid-boiling point
Flashpoint, °C	ASTM D93	Min 93	Min 120	Pensky Martens Apparatus [Closed Cup]

**Table 2**  
Properties of diesel, WGO and PO

Properties	Diesel fuel	WGO	PO
Kinematic viscosity, cST @ 40°C	3.6	17.1	1.3
Density @ 15°C, g/cm <sup>3</sup>	0.840	0.957	0.875
Lower Heating value kJ/kg	42700	35017	42800
Cetane index	45-55	54	11
Flash point, °C	74	197	52

## 2.2. Tests engine

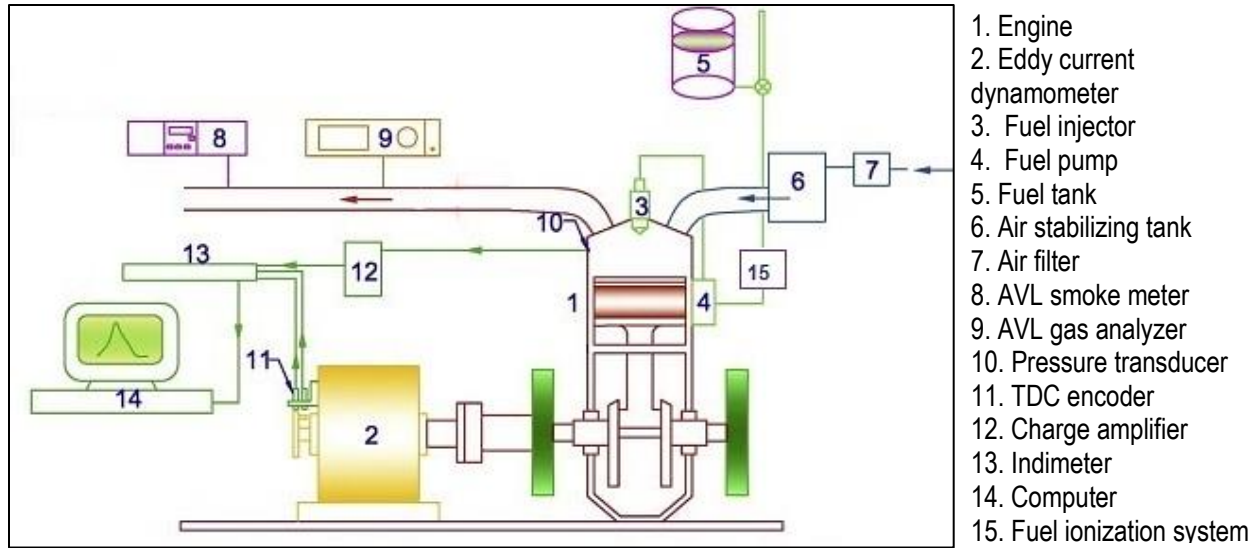
The engine used in this experimental work is a Simpson S217 twin cylinder CI engine, employed in tractors. The engine is coupled with an eddy current dynamometer and it delivers a maximum power output of 12.4 kW at a constant speed of 1500 rpm. The compression ratio (CR) of the engine is 18.5 and a hemispherical type combustion chamber is used to create the necessary swirl. The engine specifications are listed in Table 3. A mechanical type fuel injection system with a pump and 3-hole nozzle – injector assembly is used. The fuel consumption is determined manually using a burette and stopwatch; measuring the time taken to consume 10 cc of fuel.

**Table 3**  
Engine specifications

Model	Simpsons S 217 Tractor Engine
Rated Power	12.4 kW @ 1500 rpm
Type / Configuration	Vertical in-line Diesel Engine
Bore x Stroke	91.44 mm x 127 mm
No. of cylinders	2
Displacement	1670 cc
Compression Ratio	18.5:1

Fig. 1 shows the schematic diagram of the experimental setup. Engine combustion analyser model ECA 1.0.1 is used to record the combustion parameters. The in-cylinder pressure is recorded using a piezoelectric pressure transducer, Kistler HSM111A22 and the engine timing is measured through a shaft encoder which generates 360 pulses per revolution. Therefore, the data is recorded at 1 crank angle degree (CAD) intervals. The pressure data is recorded for 100 consecutive cycles and the measured values are analysed using ECA software to obtain heat release rate and ignition delay. Gaseous emissions such as carbon monoxide (CO), carbon dioxide (CO<sub>2</sub>), hydrocarbon (HC) and NO emissions are measured using AVL 5 gas analyser based on non-dispersive infrared (NDIR) technique. While CO and CO<sub>2</sub> emissions are measured in terms of percentage volume, HC and NO emissions are measured in terms of parts per million (ppm). AVL 432C smoke meter that

works based on ‘light extinction’ principle is used to measure smoke intensity in terms of percentage opacity.

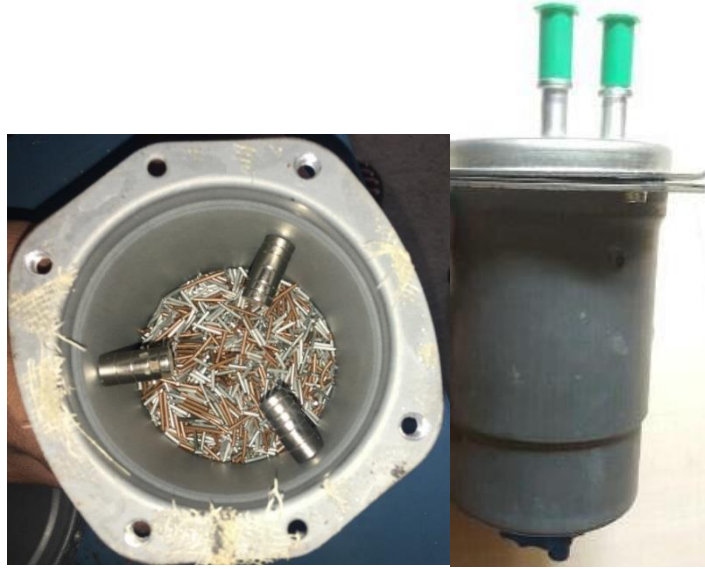


**Fig. 1.** Schematic diagram of the experimental setup

### 2.3. Fuel ionisation system

A fuel ionisation filter (FIF) using the principle of electrochemical liquid vortex ionisation is designed, manufactured and fitted before the fuel injection pump. A commercially available fuel filter is modified and converted to ionisation filter by placing three permanent magnets on the walls opposite to each other. Neodymium magnet with 2000 gauss is selected for this study. The magnets are positioned in such a way that the incoming fuel forms a vortex due to the magnetic field and becomes ionised. Metallic compounds such as copper, aluminum and galvanised iron wires are placed inside the filter, filling one third of the volume of the filter. These materials can donate electrons to the incoming fluid they come into contact with in the presence of a magnetic field. Therefore, when the fuel is atomised and sprayed into the combustion chamber, the fuel particles become more dispersed since particles with the same charge polarity repel, resulting in improved mixing with oxygen molecules in the combustion chamber [28, 29]. The FIF, shown in Fig. 2, is positioned close to the fuel injection pump since the ionised fuel was highly unstable.





**Fig. 2.** Fuel ionisation filter

#### *2.4. Experimental procedure*

All tests are performed at a constant engine speed of 1500 rpm from low to high loads, corresponding to the brake power (BP) of 3.1 kW, 6.2 kW, 9.3 kW and 12.4 kW. The fuel injection timing is kept at 23 CAD before top dead centre (BTDC) and a constant injection pressure of 200 bar is maintained throughout the experiment. All the experiments are carried out under steady state conditions without modifying the test engine. All measurements are repeated five times and the average value is used to determine the derived parameters. PO has lower cetane index and when operated as neat fuel, the engine is deemed to encounter cold start problems. Therefore, initial tests are performed with diesel as base fuel at various load conditions to attain warm-up conditions. Once the engine reaches sufficient temperature, PO is injected to avoid cold start issues. Engine performance, emission and combustion parameters of WGO and PO are measured and compared with diesel at different load conditions.

#### *2.5. Error analysis*

Uncertainty is an indicative parameter corresponding to the deviation of measured values which can be determined by uncertainty analysis. The uncertainties in experiments may arise due to instrument type, operating conditions, environmental factors and other unaccounted random

sources. The accuracy of the instruments used are identified and their corresponding uncertainty values are determined as illustrated in Table 4.

**Table 4**  
Uncertainty of various instruments and parameters

Measurement	Accuracy	% Uncertainty	Measurement technique
Load	$\pm 0.1$ kg	$\pm 0.2$	Strain gauge type load cell
Speed	$\pm 10$ rpm	$\pm 0.1$	Magnetic pickup type
Burette Fuel measurement	$\pm 0.1$ cc	$\pm 1$	Volumetric measurement
Time	$\pm 0.1$ sec	$\pm 0.2$	Manual stopwatch
Manometer	$\pm 1$ mm	$\pm 1$	Principle of balancing column of liquid
CO	$\pm 0.02$ %	$\pm 0.2$	NDIR principle
HC	$\pm 20$ ppm	$\pm 0.2$	NDIR principle
CO <sub>2</sub>	$\pm 0.03$ %	$\pm 0.15$	NDIR principle
NO	$\pm 10$ ppm	$\pm 1$	Electrochemical measurement
Smoke	$\pm 1$ % opacity	$\pm 1$	Opacimeter
EGT indicator	$\pm 1$ °C	$\pm 0.15$	K-type thermocouple
Pressure pickup	$\pm 0.5$ bar	$\pm 1$	Piezoelectric sensor
Crank angle	$\pm 1^\circ$	$\pm 0.2$	Magnetic pickup type

The uncertainty of various parameters such as total fuel consumption (TFC), brake specific energy consumption (BSEC), brake thermal efficiency (BTE) and exhaust gas temperature (EGT) are determined by the method proposed by Holman which are based on the uncertainty of the equipment used [30]. The total uncertainty of the experiment is computed by the following expression;

$$Total_{experimental} \text{ uncertainty} = \sqrt{\left\{ \begin{aligned} &(\text{uncertainty of TFC})^2 + (\text{uncertainty of BP})^2 + (\text{uncertainty of BSEC})^2 + \\ &(\text{uncertainty of BTE})^2 + (\text{uncertainty of CO})^2 + (\text{uncertainty of HC})^2 + \\ &(\text{uncertainty of NO})^2 + (\text{uncertainty of CO}_2)^2 + (\text{uncertainty of smoke})^2 + \\ &(\text{uncertainty of EGT indicator})^2 + (\text{uncertainty of pressure transducer})^2 \end{aligned} \right\}}$$

$$\frac{\text{Total experimental uncertainty}}{= \sqrt{\left\{ \frac{(1)^2 + (0.2)^2 + (1)^2 + (1)^2 + (0.2)^2 + (0.1)^2 + (0.2)^2}{(1)^2 + (1)^2 + (0.2)^2 + (1)^2} \right\}}} = 2.48\%$$

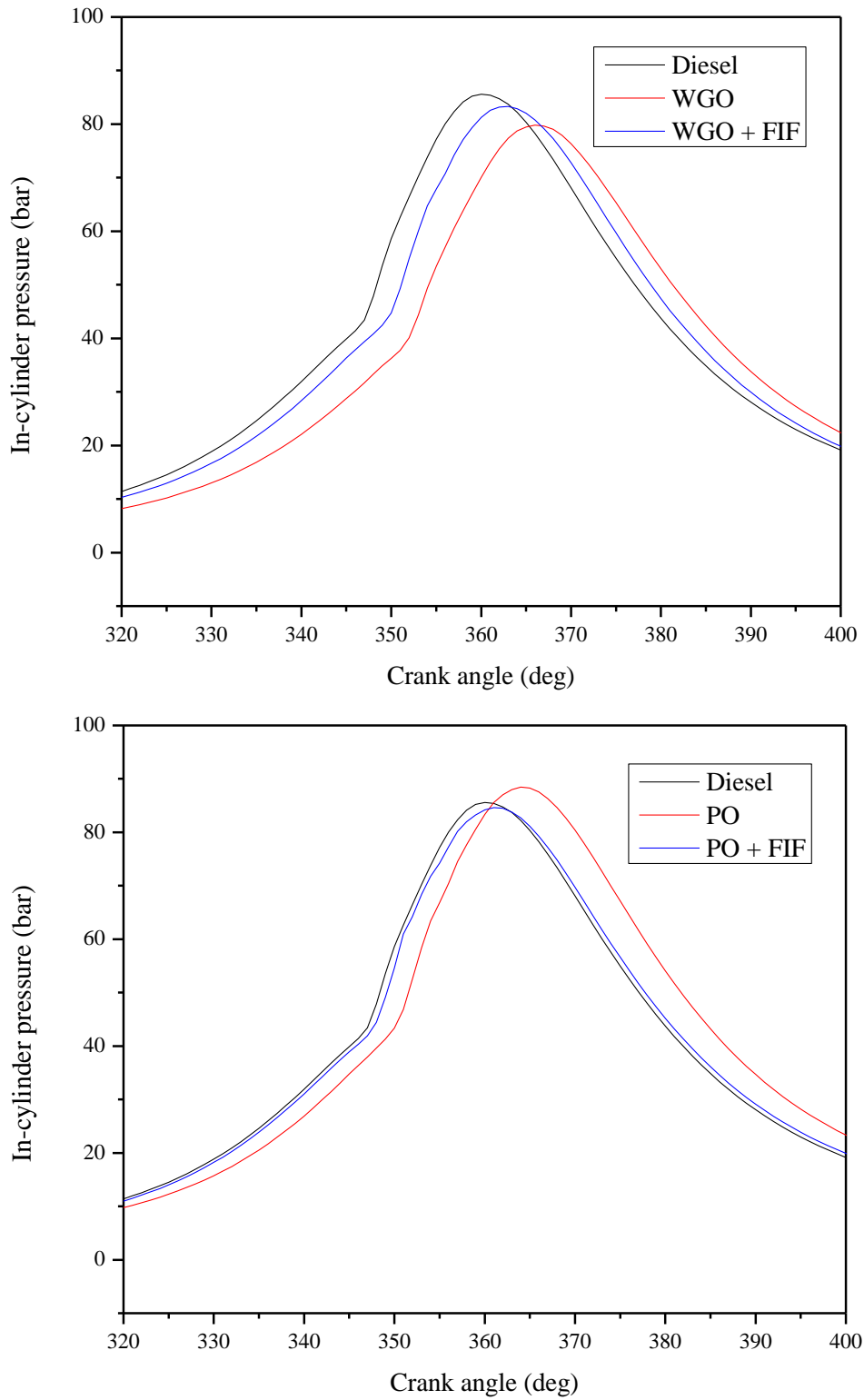
The repeatability of the results is found to be within 3% based on the Gaussian distribution.

### 3. Results and discussions

#### 3.1. Combustion characteristics

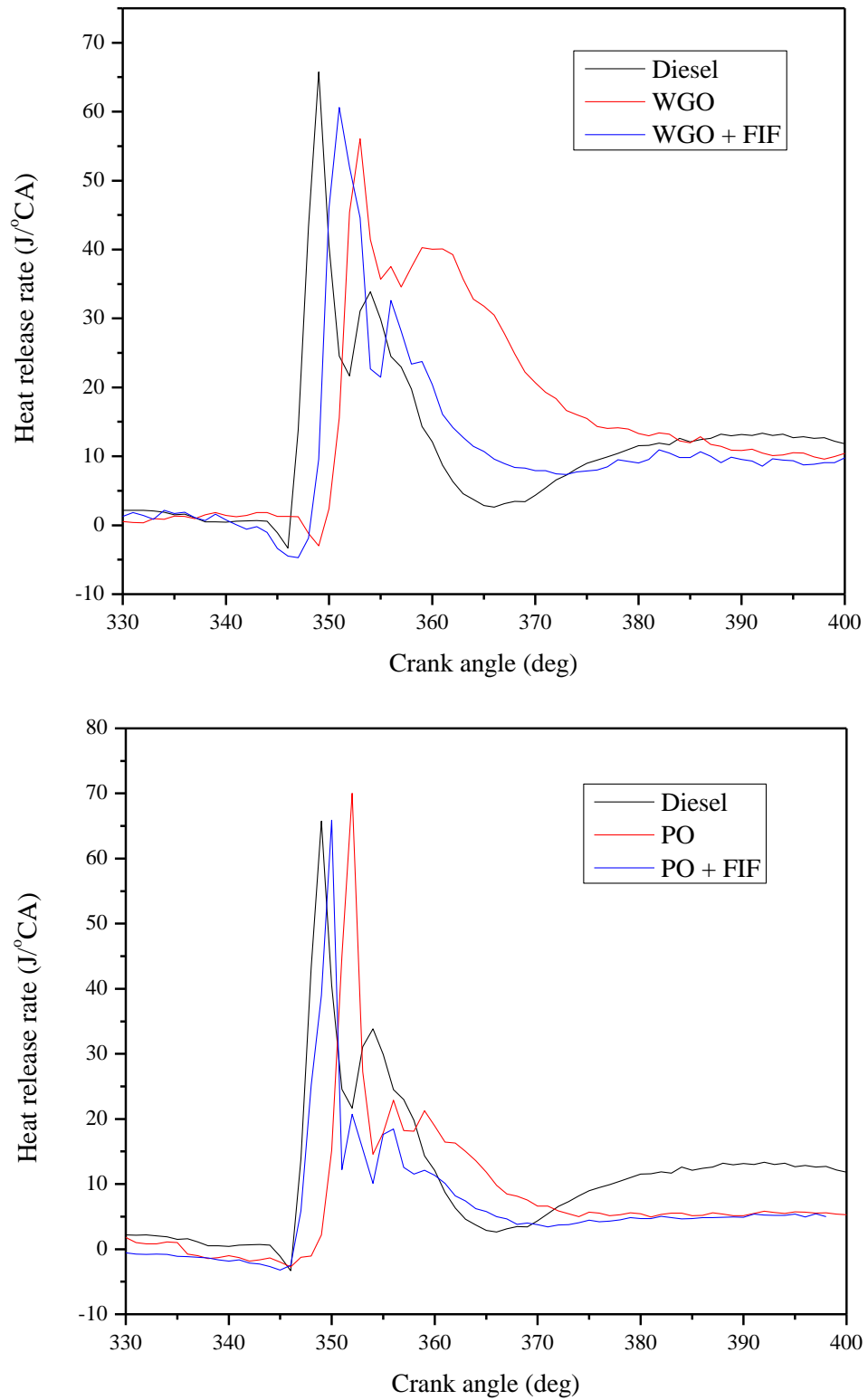
Fig. 3 illustrates the in-cylinder pressure data for diesel, WGO, WGO+FIF, PO and PO+FIF at 100% load condition. The fuel is directly injected into the combustion chamber which atomises, evaporates and mixes with air during the mixing phase. Subsequently the mixture reaches the autoignition temperature and combusts, producing mechanical work due to sudden release of energy. The timing and amplitude of the maximum pressure is mainly dependent on the amount of premixed fuel and air which is formed during the ignition delay period [31]. The results show that the peak in-cylinder pressure at 100% load condition is lower for WGO compared to diesel, 79.8 bar and 85.6 bar, respectively. The reduction in peak pressure is due to poor atomisation and evaporation of WGO compared to diesel due to higher viscosity and density, resulting in less premixed fuel and air during the ignition delay period. Similar combustion characteristics were observed with high viscous biofuels [32]. The peak in-cylinder pressure for diesel occurs at TDC while for WGO it occurs at 6 CA after top dead centre (ATDC), this shift in due to higher ignition delay with WGO. Consequently, large quantity of fuel is burnt in the later part of combustion (i.e. diffusion combustion) which results in higher EGT and lower BTE. However, with the use of the FIF the heavier droplets of WGO can be converted to ions due to the magnetisation effect, improving the atomisation and mixing characteristics during the ignition delay period [18]. The Peak in-cylinder pressure for WGO+FIF is 83.3 bar, and occurs at 3 CAD ATDC. The early occurrence of peak pressure for WGO with FIF compared to WGO is due to reduced ignition delay period owing to the quick dispersion of the ions, leading to improved combustion. In contrast, PO has a low cetane number which prolongs the ignition delay period, resulting in more profound premixed combustion and hence higher peak in-cylinder pressure due to greater fuel-air mixing [33]. The peak in-cylinder pressure for PO is 88.4 bar and occurs at 4 CAD ATDC. However, with the use of the FIF, the peak pressure of PO is reduced to 84.6 bar and occurs earlier, this may be

due to earlier SOC caused by the ionisation of PO. The results demonstrate that FIF has minimal effect on low viscous biofuels compared to high viscous biofuels.



**Fig. 3.** In-cylinder pressure data at 100% load

Fig. 4 depicts the heat release rate data for diesel, WGO, WGO+FIF, PO and PO+FIF at 100% load condition.

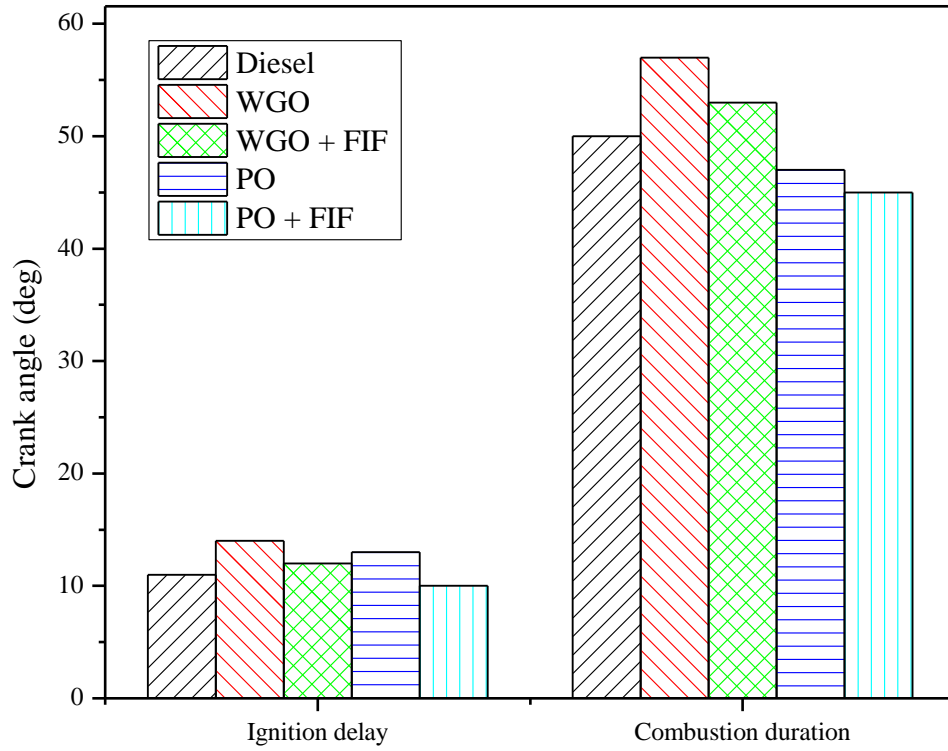


**Fig. 4.** Heat release rate data at 100% load

In general, high viscous biofuels exhibit inferior combustion characteristics due to poor atomisation, evaporation and mixing with air during the ignition delay period [34]. The peak heat release for diesel and WGO are 65.7 J/°CA and 56 J/°CA, respectively. The peak heat release for WGO occurs later in the cycle and shows more diffusion combustion compared to diesel, this is mainly due to poor atomisation, evaporation and air-fuel mixing resulting in longer ignition delay and more fuel burning later in the cycle. This has an adverse effect on engine performance and leads to higher soot emission due to the presence of fuel-rich combustion. Kasiraman et al. [35] investigated cashew nut shell oil in a CI engine and reported similar findings. The peak heat release rate for WGO+FIF is 60.6 J/°CA and occurs earlier in the cycle compared to WGO due to ionisation of heavy WGO fuel, enhancing fuel atomisation, evaporation and mixing processes during the ignition delay period. Consequently, less diffusion combustion occurs which reduces the soot emission and improves the engine performance. On the other hand, low cetane number of PO leads to longer ignition delay resulting in the formation of more premixed fuel-air mixture [36]. As a result, peak heat release of PO is 70 J/°CA and occurs later in the cycle compared to diesel. However, unlike WGO, PO exhibits better combustion characteristics due to more profound premixed combustion. The ignition delay of PO+FIF is shorter resulting in reduced peak heat release rate of 65.9 J/°CA.

Fig. 5 shows the ignition delay and combustion duration for WGO, PO with and without FIF compared to diesel at 100% load. Ignition delay occurs due to inherent physical and chemical properties of the fuel referred to as physical and chemical delays, respectively. The former is due to the time required for the fuel to atomise, evaporate and mix with air to form a combustible mixture while the latter is dependent on the cetane number of the fuel which promotes autoignition of the premixed mixture [37]. Ignition delays for diesel and WGO are 11 and 14 CAD at 100% load. Prolong ignition delay of WGO is due to physical delay caused by high viscosity and density of the fuel resulting in poor atomisation, evaporation and fuel-air mixing. Agarwal and Dhar [38] also observed a longer ignition delay with karanja oil. The ignition delay for WGO+FIF is reduced to 12 CAD, this is attributed to ionisation effect caused by the magnets promoting early onset of combustion. The ignition delay for PO is 13 CAD which is longer compared to diesel. Although physical delay with PO is lower compared to diesel due to lower viscosity and density, the chemical delay is significantly longer due to low cetane number. However, ignition delay of

PO+FIF is reduced to 10 CAD, this is due to further reduction in physical delay due to ionisation of fuel particles.

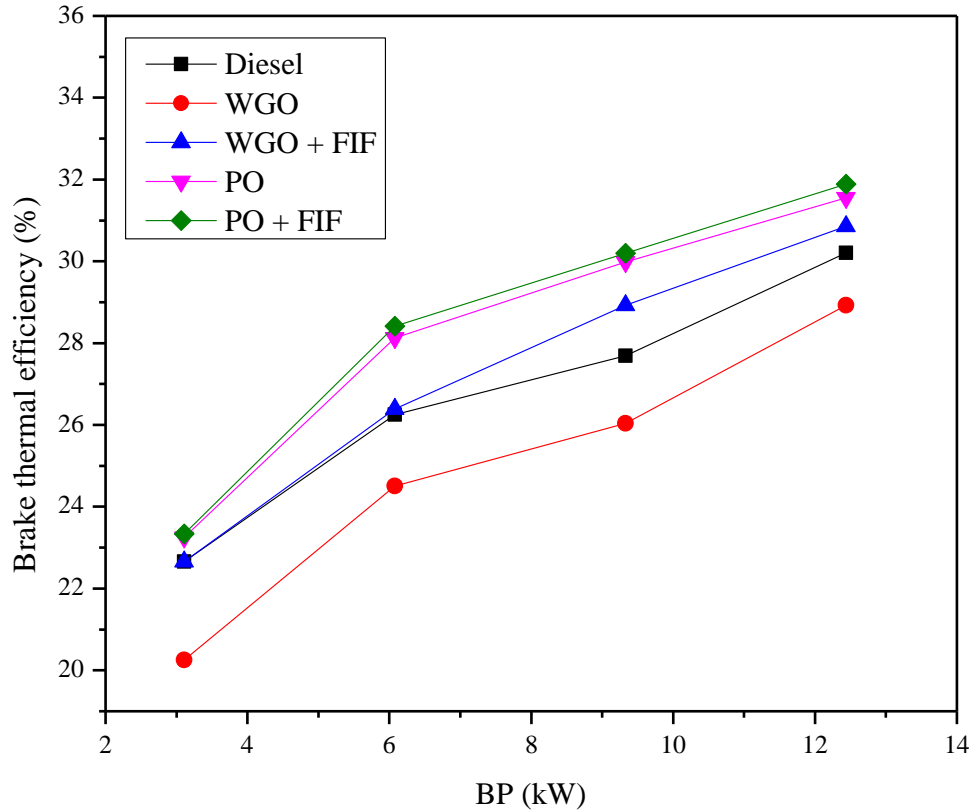


**Fig. 5.** Ignition delay and combustion duration characteristics at 100% load

Combustion duration is calculated as the start of combustion and 90% mass fraction burnt. Combustion durations for diesel and WGO are 50 CAD and 57 CAD at 100% load, respectively. Longer combustion duration with WGO is due to late start of combustion for WGO, as evident in Fig. 4. The high viscosity and density of WGO lead to poor fuel atomisation and less premixed mixture formation which results in more diffusion combustion as shown in Fig. 4. Consequently, higher EGT and lower BTE are obtained with WGO compared to diesel. Enhancement in combustion is observed with FIF due to improved air-fuel mixing. Combustion duration for WGO+FIF is 53 CAD, shorter compared to WGO at 100% load. Combustion durations for PO and PO with FIF are 47 and 45 CAD at 100% load which is shorter compared to diesel due to improved combustion as a result of better atomisation and mixing with air owing to lower viscosity. The results indicate that that FIF has minimal impact on combustion duration.

### 3.2. Engine performance

Brake thermal efficiency is the ratio of brake power to fuel energy indicating the conversion efficiency of fuel energy to mechanical power output. Fig. 6 shows the variation of BTE for diesel, WGO, WGO+FIF, PO and PO+FIF at different load conditions.



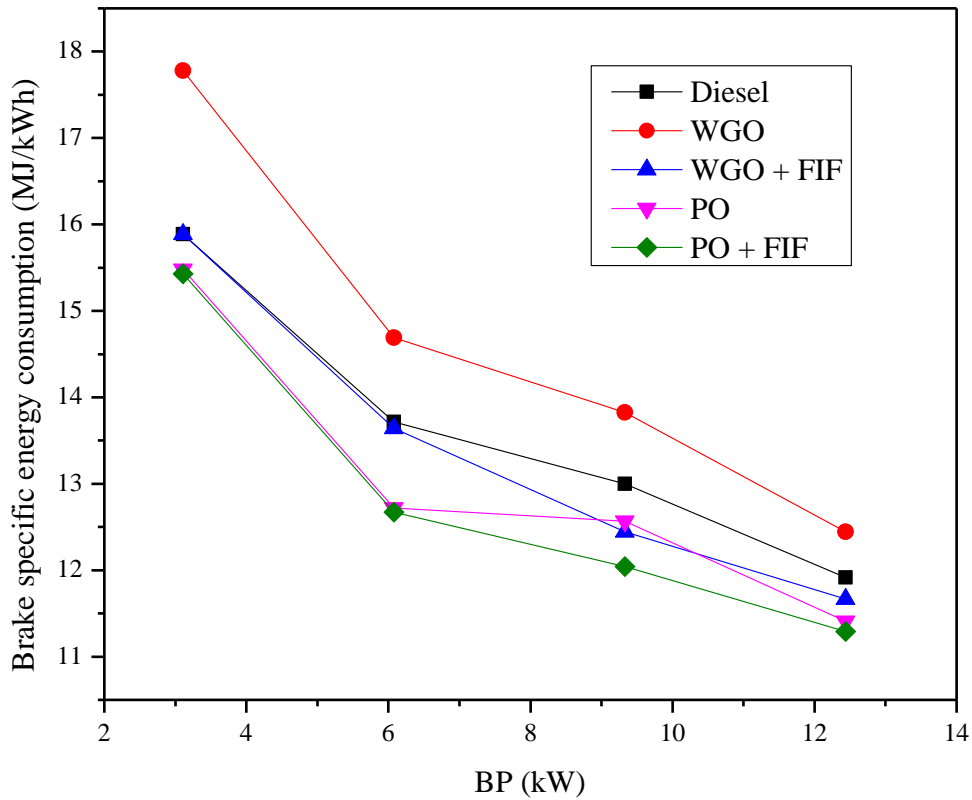
**Fig. 6.** Brake thermal efficiency at different load conditions

BTE increases with increase in load due to reduction in heat loss and friction. At 100% load, BTEs for diesel and WGO are 30.2% and 28.9%, respectively. BTE is dependent on the amount of heat release during the premixed combustion phase [39], thus the reduction in BTE for WGO is attributed to poor mixture formation due to higher viscosity and density. This is confirmed by the heat release rate data shown in Fig. 4. BTE is improved to 30.85% for WGO+FIF compared to 28.9% for WGO at 100% load. BTE is increased with the use of FIF due to improved mixing of fuel and air in the combustion chamber owing to ionisation effects. The ionised cationic molecules (negatively charged) repel inside the combustion chamber causing the molecules to disperse and mix with oxygen molecules [18]. This is also evident in Fig. 4 where WGO+FIF resulted in higher



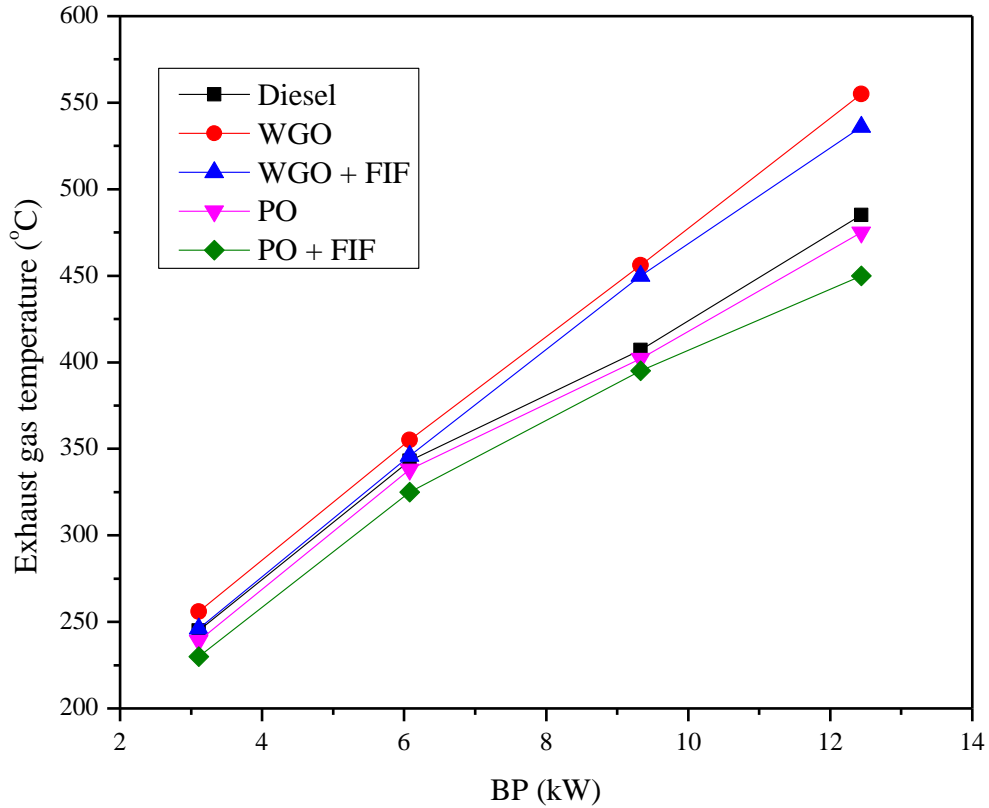
premixed combustion. The same trend is observed at other load conditions. BTEs for PO and PO+FIF are 31.55% and 31.88% at 100% load, respectively. BTE is improved with PO compared to diesel at all load conditions. This is mainly attributed to better atomisation, dispersion, evaporation and mixing characteristics of low viscosity biofuels [24]. However, it is evident that ionisation of such biofuels has minimal impact on BTE.

Fig. 7 shows BSEC for diesel, WGO, PO with and without FIF at different load conditions. At 100% load, BSEC for diesel, WGO and WGO+FIF are 11.91 MJ/kWh, 12.44 MJ/kWh and 11.66 MJ/kWh, respectively. Higher BSEC for WGO is attributed to poor combustion characteristics due to unfavourable properties such as higher viscosity, density and low heating value. However, this trend is reversed with the use of FIF due to improved fuel atomisation, evaporation and fuel-air mixing. Previous studies reported that BSEC of low viscous vegetable oil is lower compared to diesel [33, 35, 36]. BSEC for PO and PO+FIF are 11.4 MJ/kWh and 11.28 MJ/kWh at 100% load, respectively. BSEC for PO is slightly lower compared to diesel due to lower viscosity and density. In contrast to WGO, ionisation has almost no effect on BSEC of PO.



**Fig. 7.** Brake specific energy consumption at different load conditions

Fig. 8 shows exhaust gas temperature for diesel, WGO, PO with and without FIF at different load conditions. EGTs for diesel, WGO and WGO+FIF are 485°C, 555°C and 536°C respectively. EGT for WGO is higher compared to diesel which is an indication of higher thermal loss due to slower combustion process leading to extended combustion phasing [40]. As seen in Fig. 4, the diffusion phase for WGO occurs later in the cycle compared to diesel which causes more heat rejection by exhaust gases reducing the brake power output. However, EGT for WGO+FIF is lower compared to WGO at all load conditions due to shorter ignition delay resulting in earlier onset of diffusion phase. EGTs for PO and PO+FIF are 475°C and 450°C, respectively. Although, ignition delay for PO is longer compared to diesel, the combustion phasing is less due to faster combustion of PO [36] which results in reduced EGT and higher BTE. The use of FIF leads to earlier onset of combustion compared to PO, reducing EGT and the thermal loss.

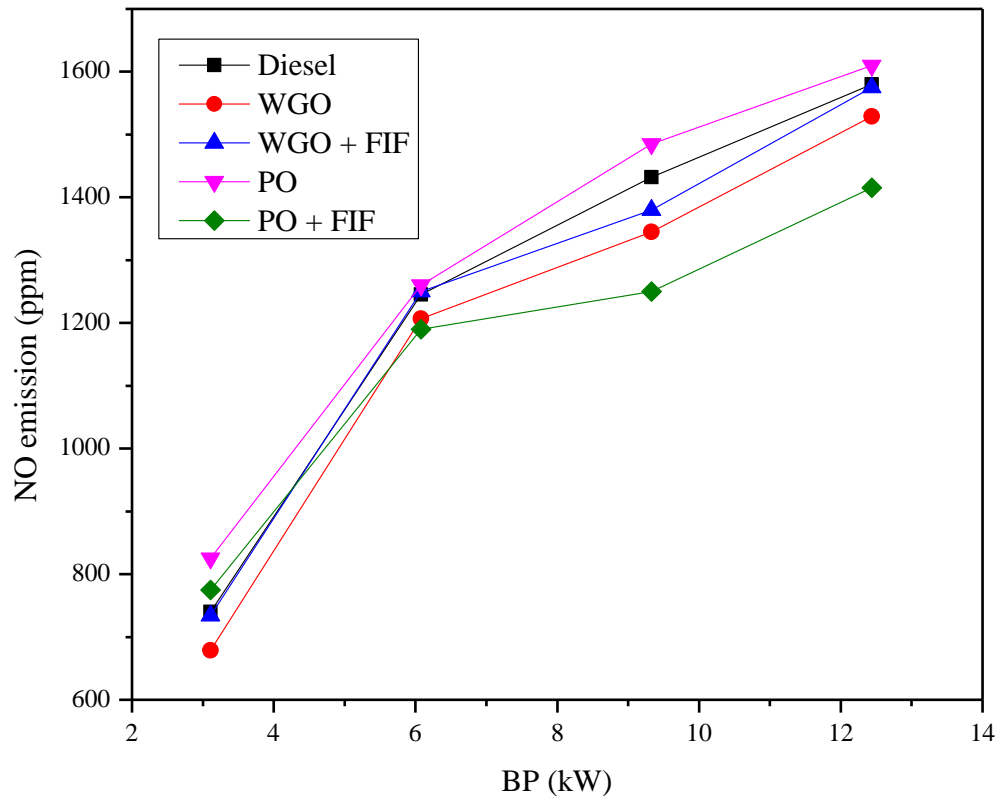


**Fig. 8.** Exhaust gas temperature at different load conditions

### 3.3. Emissions

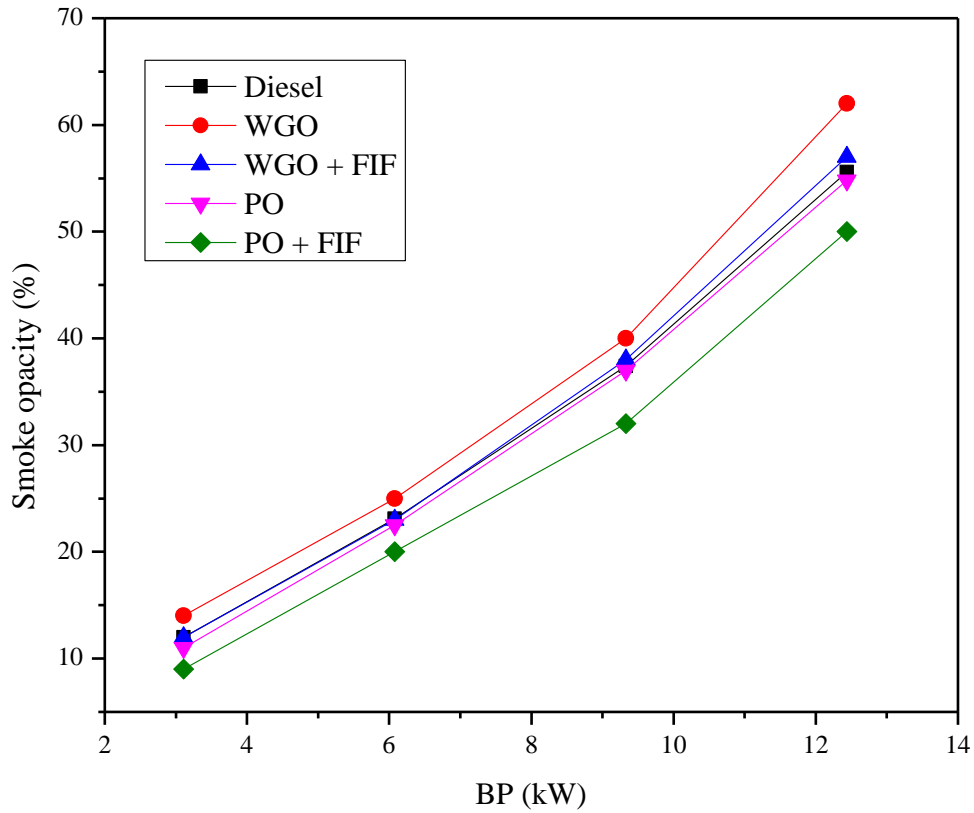
Fig. 9 illustrates NO emission for diesel, WGO, PO with and without FIF at different load conditions. NO emission is formed due to high combustion temperature and presence of oxygen.

NO emission in CI engines is primarily formed during the premixed combustion phase due to higher combustion temperature [41]. NO emission increases with increase in load due to higher combustion temperature at higher load conditions. At 100% load, NO emission for diesel, WGO and WGO+FIF is 1580, 1529 and 1575 ppm. NO emission for WGO is less compared to diesel at all the load conditions, due to less profound premixed combustion phase for WGO. However, with FIF the combustion is improved resulting in higher combustion temperature and higher NO emission formation. NO emission is generally higher for low viscous biofuels due to low cetane number. Low cetane number leads to longer ignition delay resulting in higher heat release rate during the premixed combustion phase. NO emission for PO and PO+FIF are 1610 ppm and 1415 ppm, respectively. Higher NO emission is formed with PO due to longer ignition delay. However, NO emission is reduced by approximately 12% for PO+FIF compared to PO. The reduction in NO emission with FIF is due to reduced ignition delay. This leads to less profound premixed combustion thus lowering the combustion temperature.



**Fig. 9.** NO emission at different load conditions

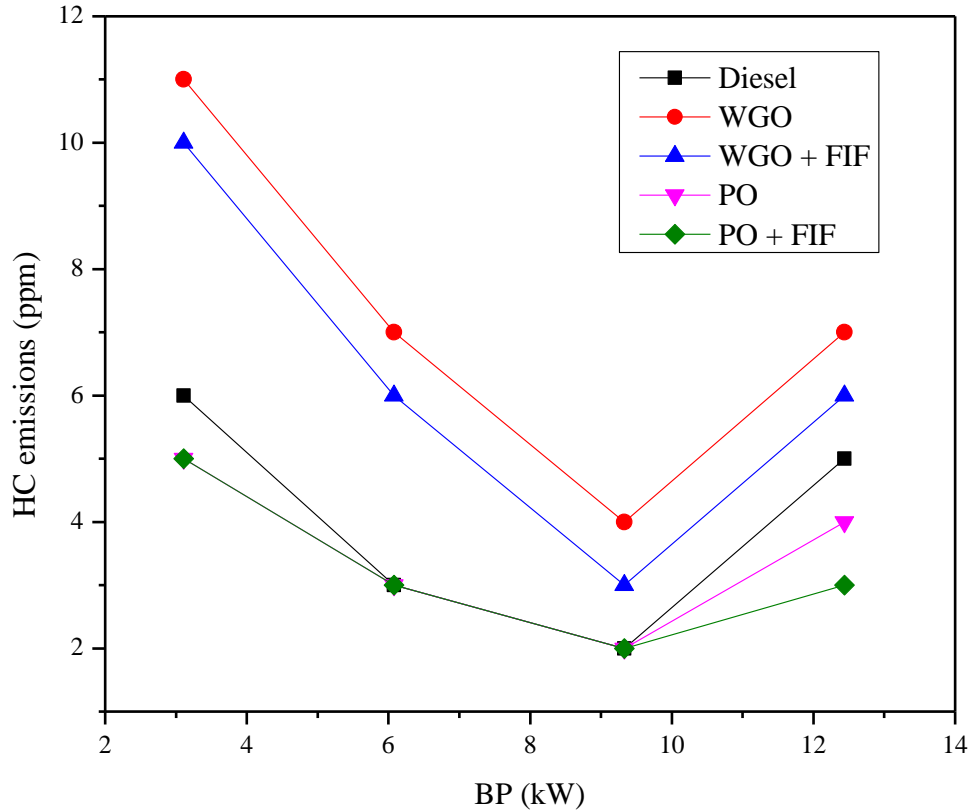
Fig. 10 shows smoke opacity for diesel, WGO, PO with and without FIF at different load conditions. Soot emission is formed during the diffusion combustion and increases with increase in load due to increase in fuel injection quantity. At 100% load, smoke opacity for diesel and WGO are 55.6% and 62% respectively. Higher smoke emission for WGO is due to more profound diffusion combustion due to high viscosity and density as shown in Fig. 4 [35]. The smoke emission for WGO+FIF is reduced to 57% at 100% load due to enhanced atomisation, evaporation and fuel-air mixing resulting in less diffusion combustion. The smoke emission for PO and PO+FIF are 54.8% and 50% at 100% load, respectively. Smoke emission is less for low viscous biofuels due to improved atomisation and better mixing with air. FIF implementation aids in further enhancement of fuel-air mixing resulting in further reduction of smoke emission.



**Fig. 10.** Soot emission at different load conditions

Formation of HC emissions in CI engines are mainly due to engine operating conditions, fuel properties, spray characteristics and air-fuel ratio [42]. HC emissions are less in CI engine compared to SI engine due to leaner combustion. Fig. 11 shows HC emissions at different load conditions for the test fuels. At 100% load, HC emission for diesel, WGO and WGO+FIF are 5

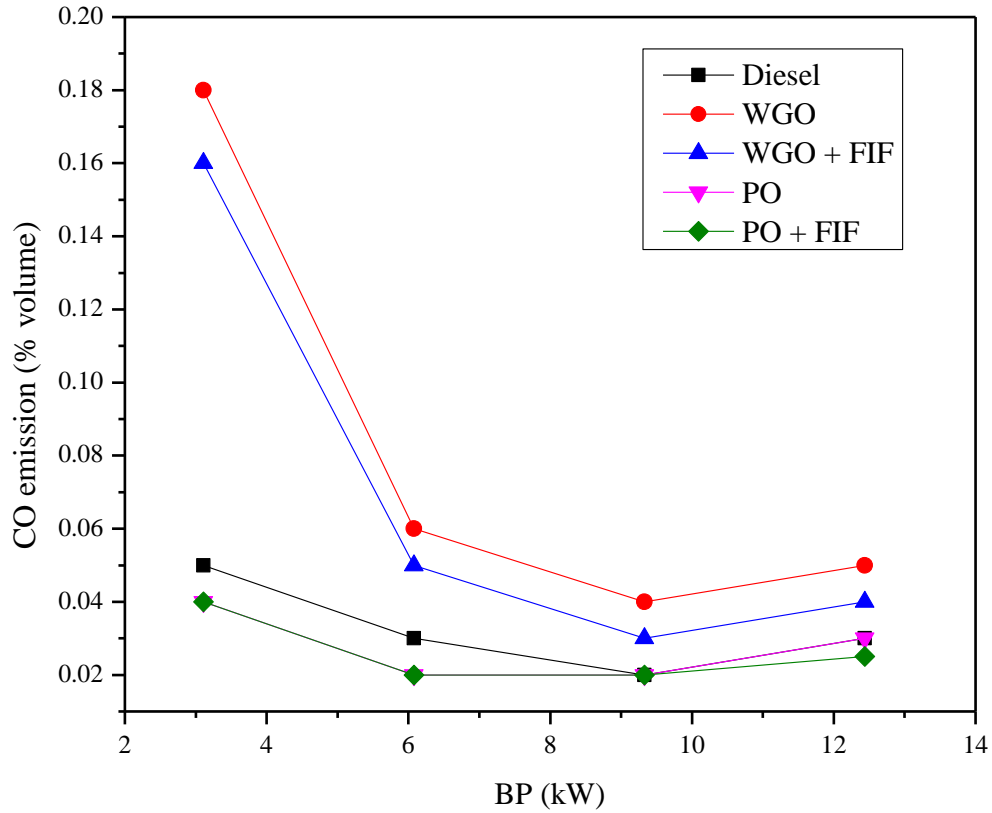
ppm, 7 ppm and 6 ppm, respectively. Higher HC emission for WGO is due to poor spray characteristics owing to high viscosity of the fuel. This leads to lower combustion temperature masking the oxidation process of HC. At low load conditions, HC emission is approximately 80% higher for WGO compared to diesel. This is due to very low combustion temperature at low loads and poor fuel properties. With FIF, HC emissions is slightly reduced due to improved fuel atomisation which leads to improved combustion resulting in oxidation of HC at high temperatures. At 100% load, HC emission for PO and PO+FIF are 4 ppm and 3 ppm due to superior fuel atomisation, evaporation and fuel-air mixing. HC emission is slightly lower for PO due to improved combustion and lower C-H ratio in the fuel structure. The results demonstrate that HC emission reduces with FIF at all load conditions.



**Fig. 11.** HC emission at different load conditions

Fig. 12 shows the CO emission at different load conditions for the test fuels. At 100% load, CO emission for diesel, WGO and WGO+FIF are 0.03%, 0.05% and 0.04%, respectively. Higher CO emission for WGO is due to higher viscosity of WGO which leads to inefficient fuel-air mixing favoring CO emission formation during the diffusion combustion phase where the combustion

temperature is lower. CO emission is reduced with FIF due to improved fuel atomisation and dispersion resulting in more complete combustion with less intermediate CO emission formation. CO emission for PO and PO+FIF are 0.03% and 0.025% at 100% load, respectively. Better fuel atomisation due to lower viscosity reduces CO emission which is further reduced with the FIF.



**Fig. 12.** CO emission at different load conditions

#### 4. Conclusions

In this paper the effect of high and low viscosity biofuels on combustion and emissions were investigated in a twin cylinder CI engine. Wheat germ oil and pine oil were tested at different load conditions as diesel fuel surrogates with high and low viscosities, respectively. WGO suffered from poor atomisation, evaporation and fuel-air mixing due to its high viscosity which resulted in inferior combustion characteristics. In contrast, PO benefited from improved atomisation and mixing due to lower viscosity, however, its low cetane number resulted in longer ignition delay and higher NO emission. A bespoke fuel ionisation system with a permanent magnet was fitted before the fuel pump to ionise the incoming fuel prior to injection. The tests were carried out with

fixed injection timing of 23 CAD BTDC and at constant engine speed of 1500 rpm. At 100% load, BTE reduced from 30.2% to 28.92% for WGO compared to diesel and improved to 30.85% for WGO+FIF. Although, BTE for PO was higher compared to diesel due to lower viscosity, unlike WGO FIF had minimal impact on PO. NO emission was lower for WGO compared to diesel due to less profound premixed combustion caused by poor atomisation and mixing. However, NO emission increased with FIF due to improved premixed mixture formation which resulted in higher temperature combustion. NO emission for PO was higher compared to diesel due to longer ignition delay caused by low cetane number and reduced with FIF due to improved atomisation and mixing resulting in shorter ignition delay period. Higher HC, CO and soot emissions were observed with WGO compared to diesel, however, they reduced with FIF due to improved combustion characteristics. HC, CO and soot emissions were less with PO due to lower viscosity and further reduced with FIF due to ionisation of fuel. Heat release rate and peak in-cylinder pressure was less and occurred later in the cycle for WGO compared to diesel and improved with FIF due to improved fuel atomisation and mixture formation during the ignition delay period. Longer ignition delay for PO increased the heat release rate and peak in-cylinder pressure which was slightly reduced with FIF.

Use of high and low viscosity biofuels in CI engines can adversely affect engine performance due to poor mixture formation and longer ignition delay. The use of FIF addresses the challenges associated high and low viscosity biofuels. The problem associated with high viscosity is resolved by fuel ionisation which improves fuel atomisation and fuel-air mixing while the ignition delay is reduced with low viscosity biofuels due to shorter physical delay which in turn reduces NO emission. The proposed method of fuel ionisation can be easily retrofitted in existing and future diesel vehicles eliminating the need for complex and expensive transesterification processes.

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